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Experimental investigation of a two-stage oil-free domestic Air/Water heat pump prototype powered by an oil-free high-speed twin-stage radial compressor rotating on gas bearings

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Experimental investigation of a two-stage oil-free domestic Air/Water heat pump prototype powered by an oil-free high-speed twin-stage radial compressor rotating on gas bearings

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ABSTRACT

Domestic heat pumps have been identified as a key-technology to decrease the energy consumption of the domestic sector, which represents 29% of the world final energy consumption. Space & water heating represent more than two third of the domestic sector energy consumption in many countries. Next generation domestic heat pumps powered by oil-free high-speed radial compressors are expected to (a) be more efficient at rating operating points and in partial loads, (b) to match better the energy demand characteristics, (c) to be more compact and lighter, (d) more silent, and (e) to need lower refrigerant charges. As a first step towards the development of such advanced heat pumps, an oil-free twin-stage Air/Water domestic heat pump prototype powered by an oil-free high-speed twin-stage radial compressor has been developed and tested. The heat pump layout corresponds to a two-stage heat pump cycle with an open economizer. The two main functions of the economizer are (a) to separate the liquid from the gas before the second-stage compressor inlet and offer a proper mixing process, and (b) to store the liquid refrigerant not located in the heat exchangers at a given time. This key-component has been coupled with the compression unit and improved through incremental experimental steps. The heat pump driven by an oil-free twin-stage radial compressor has been successfully tested at the rating operating point A-7/W35. The performance reached with the prototype is considered very promising and constitutes a breakthrough in the domain. This article describes the experimental setup, the Operating Point (OP) A-7/W35, the methodology applied for data analysis, followed by the results. The issues limiting the performance of the prototype are also identified and briefly documented.

1. INTRODUCTION

The worldwide domestic energy consumption almost reaches a third of the global consumption and is mostly provided by fossil fuels (International Energy Agency, 2008, p. 17). Moreover, in the IEA19 countries, space heating remains the most important energy use, responsible for 53% of household final energy consumption (International Energy Agency, 2008, p. 46). Both space and water heating result in an almost stable consumption of about 70% of the energy used in the domestic sector over the last 25 years (International Energy Agency, 2008, p. 46). Improved domestic heat pumps are identified as a key-technology to decrease the energy consumption of this sector. Future heat pump systems should be (a) more efficient, (b) more compact, use (c) less material, make (d) less noise, and have a (e) low impact on the environment or the climate. Zehnder (2004) and Barbouchi (2007) demonstrated in their work the importance of the exergy losses in the compression process. As a consequence, in order to significantly improve the performance of the domestic heat pumps, the compression process efficiency has to be improved as much as possible. Therefore, to improve the domestic heat pump performance, a technological step on the compression device is necessary. This served as a key motivation for the radial compressor development performed by Schiffmann (2008).

2. A TWO-STAGE HEAT PUMP CYCLE WITH AN OPEN ECONOMIZER

The heat pump prototype described and tested in this work consists of a two stage cycle with an open economizer, as illustrated in the thermodynamic diagrams presented in Fig. 3, and is driven by an oil-free twin-stage radial compressor supported on gas-lubricated bearings. It is the first domestic size heat pump successfully tested worldwide with a 6kW radial compressor with 2 stages. The cycle is equipped with an original economizer, developed for this application, including an efficient liquid/gas separator, and combined with the volute of the compression unit.

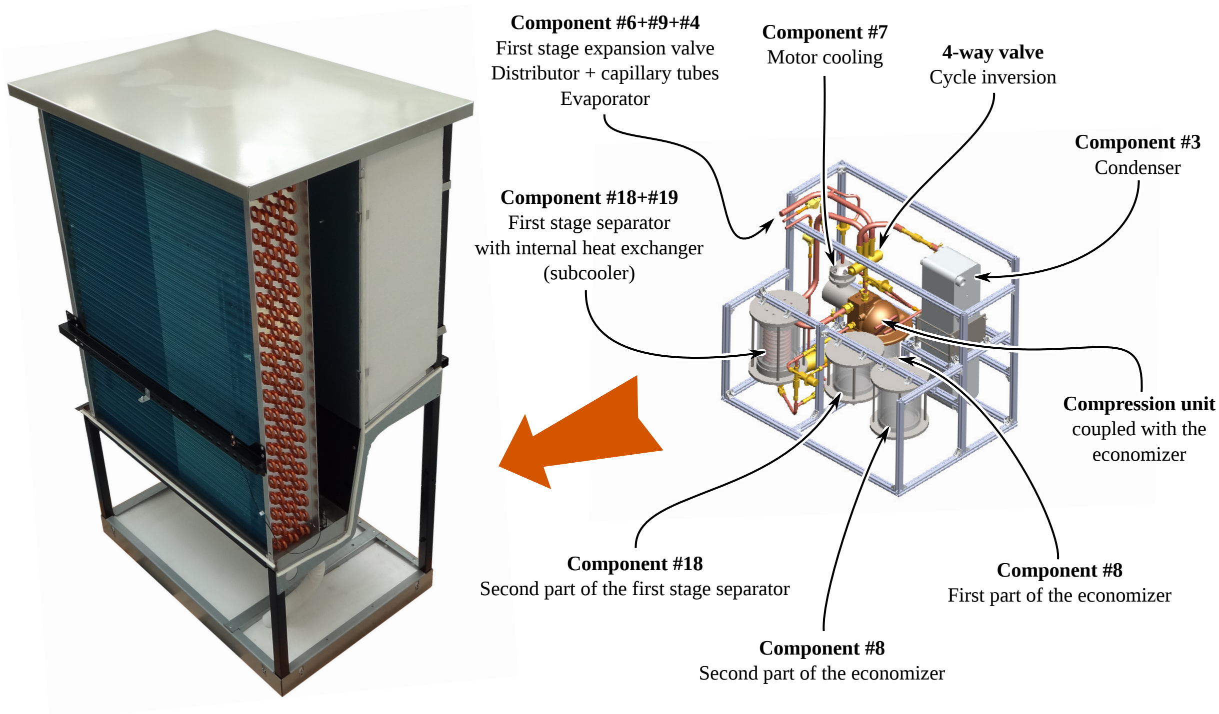


Figure 1: Topology of the twin-stage Air/Water heat pump Prototype (AWP)

Two-stage heat pump cycles have proven to reach higher performance than single-stage cycles (Favrat et al., 1997; Zehnder, 2004). However, there are many ways to perform two-stage heat pump cycles. Schiffmann & Favrat (2009) concluded that the open economizer layout is the most interesting one, when taking into account the radial compressors characteristics and limitations. The heat pump layout is detailed in Fig. 2.

The goal for this heat pump prototype was to demonstrate the potential of an oil-free radial variable-speed multi-stage compressor unit for domestic heat pumps applications. It has been designed to be integrated into an existing single-stage Air/Water heat pump housing. The selected Air-Water heat pump is designed for cold climates. Hence, their heat pump circuits and components are located below the air ducting, this allows to keep the air ducting above snow level with minimum footprint. The heat pump prototype was also intended to demonstrate the simplicity of the heat pump layout selected, resulting in positive effects on the manufacturing cost. As a consequence of those design choices, the heat pump layout topology had to fit into the space available below the air ducting, as illustrated in Fig. 1.

The AWP is equipped with a glass-made separator and economizer to allow the monitoring and understanding of the refrigerant distribution. To prevent liquid management issues during the tests at the laboratory, the inlet separator and the economizer should be able to contain a good share of the refrigerant charge. In order to decrease the impact of control mistakes during the experiments, the volumes of the first stage compressor inlet separator and of the economizer have been increased by a factor two by duplicating them.

Gas/liquid separation issues were encountered in the economizer during the experiments, therefore the design of the gas/liquid separation parts in the economizer have evolved significantly during the experimental investigation. The successive versions are detailed in Fig. 4. The data presented in this work has been generated with the AWP equipped

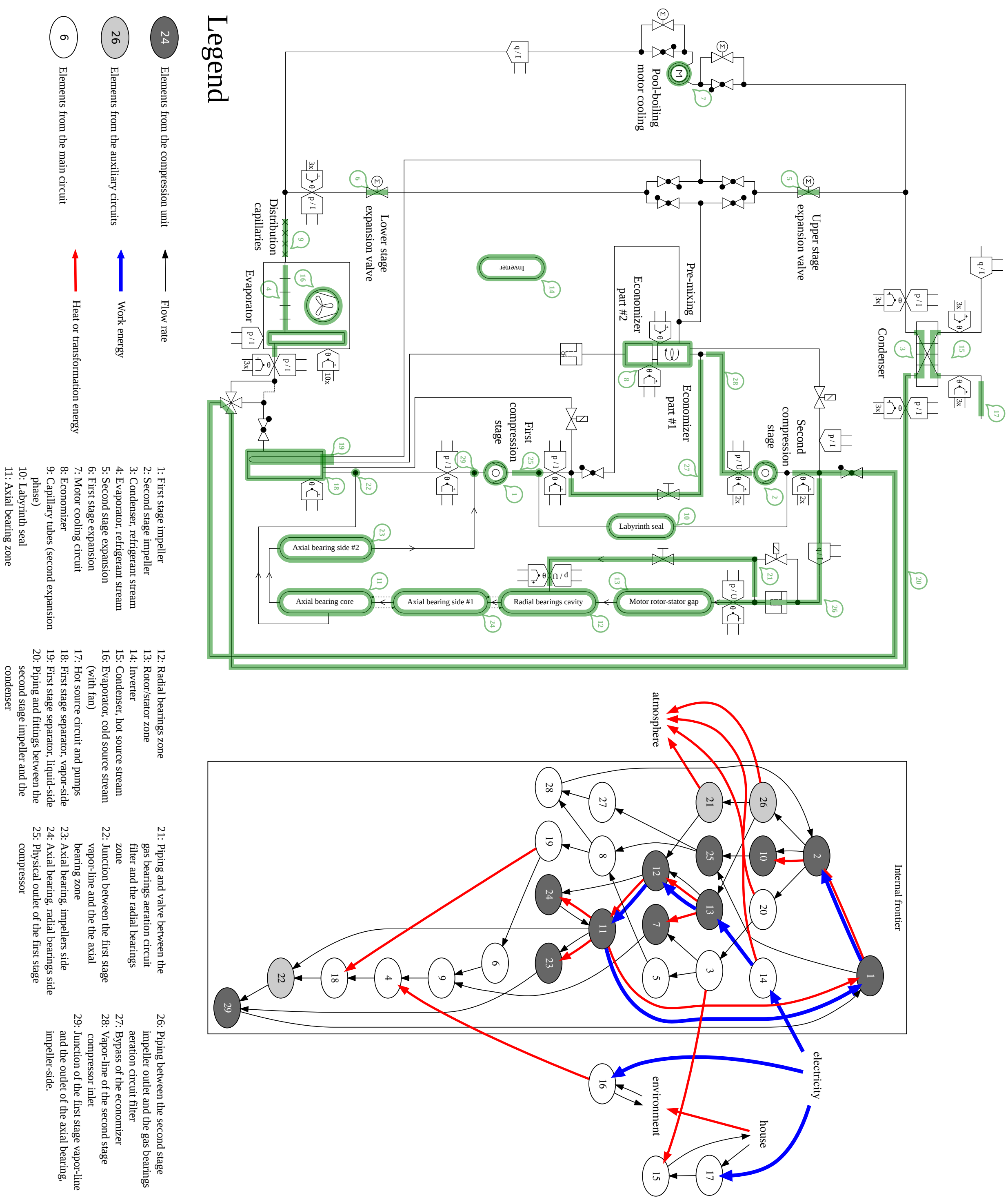


Figure 2: Layout and model of the AWP

with the version #5 of the economizer, which is able to contain the refrigerant charge and to perform an appropriate gas/liquid separation.

3. A TWIN-STAGE RADIAL COMPRESSOR

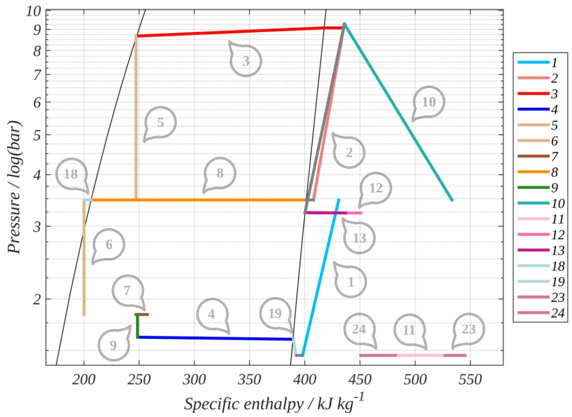
The AWP is equipped with a compression unit containing the two radial compression stages, mounted on the same shaft, as illustrated on Fig. 5. The unit maximum rotation speed is 180 krpm for a maximum shaft power of 6 kW. None of the compression stages are equipped with inlet guides and vaneless diffusers are being used for maximum compressor range. The two impellers have been designed in order to work together, using the methodology developed by Schiffmann & Favrat (2010). The efforts on the shaft are recovered by a two sided inward pumping spiral groove thrust bearing and 2 herringbone grooved journal bearings that are both lubricated with vapor phase refrigerant fluid. The compressor unit is therefore a completely hermetic system. The unit needs 3 types of auxiliary refrigerant circuits: a motor-cooling circuit, a gas bearings aeration circuit, and bypass circuits. The unit is equipped with a pool-boiling motor cooling. A drawing of the compression unit is presented in Fig. 5. Bypass circuits are mandatory auxiliary circuits in heat pumps integrating the radial compression units. In the AWP, each compression stage is equipped with a bypass circuit consisting of a manual needle valve, used to set the characteristics of the bypass, and a solenoid valve for rapid bypass activation. Being able to open or close the bypass circuits for each compression stage separately does not make them independent, however. An other solution, apparently better, tested on another prototype, was to have a fast moving electric needle valve mounted on a circuit that bypasses both compression stage together (Carré, 2015, p. 73).

4. MODELING OF THE HEAT PUMP PROTOTYPE

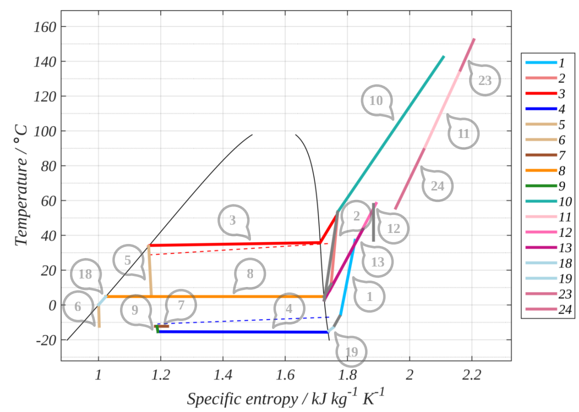
The compression unit is particularly compact and difficult to instrument: pressure, temperature, and flow rate measurements inside the unit are difficult to perform. Moreover, refrigerant flow rates in the condenser and the evaporator were not measured, as the prototype had been designed with compactness and performance in mind. Consequently, a reconciliation of the experimental data through a detailed model is necessary to understand what happens inside the compression unit and to know the characteristics of the refrigerant flow at any location of the cycle. The modeling methodology selected here implies the separation of the the heat pump layout in components, which are used to write the mass flow rate and energy balance equations. A component is a network, as intended by Borel & Favrat (2010, p. 24–25), and corresponds to an element where flows enter and leave, partially (if there is accumulation) or totally, and that can exchange energy with normal components. When those components are defined, the layout view becomes the model view (presented in Fig. 2), where the components are linked together with their mass and energy exchanges. The links between the components allow to write the set of equations governing the system. Each components is described by a mass flow rates balance and an energy balance. The equations are solved beginning with the mass flow rates equations, which have the priority. When no more equations can be solved, fitting parameters are added with the introduction of new relations linking variables together. When all the mass flow rates are determined, the energy balance equations are written with the same method. Next, the model is implemented in a modeling software (here, the MATLAB from MathWorks) and the set of equations is solved with a solver (here, the function *fmincon*). The solver tries to fit the parameters that were added, in order to get the lowest objective function value. The objective function is based on the sum of each balance equations. When the equations are equal to zero, the objective function is equal to zero. REFPROP 9.0 (Lemmon et al., 2011) is used to compute the fluid properties. Following this procedure, the model is built with 29 components; 11 of them describe the compression unit itself. The model is presented in Fig. 2.

5. EXPERIMENTAL RESULTS

5 stable OP have been documented with the AWP (Carré, 2015, p. 46), notably the OP A-7/W35, which is detailed in this section. This OP allows to compare the AWP performance with industrial products using conventional technology, even if the comparison is not very accurate, as the prototype tests were not strictly performed in the conditions imposed by the standards relative to performance measurement of heat pumps with electrically driven compressors for space heating & cooling applications (AFNOR, 2011a,b,c,d, 2012). Notably, the fan and pump electrical powers were not taken in account (AFNOR, 2011c, sec. 4.1.4 and 4.1.6), the condenser water temperature difference was of 6.84 K,

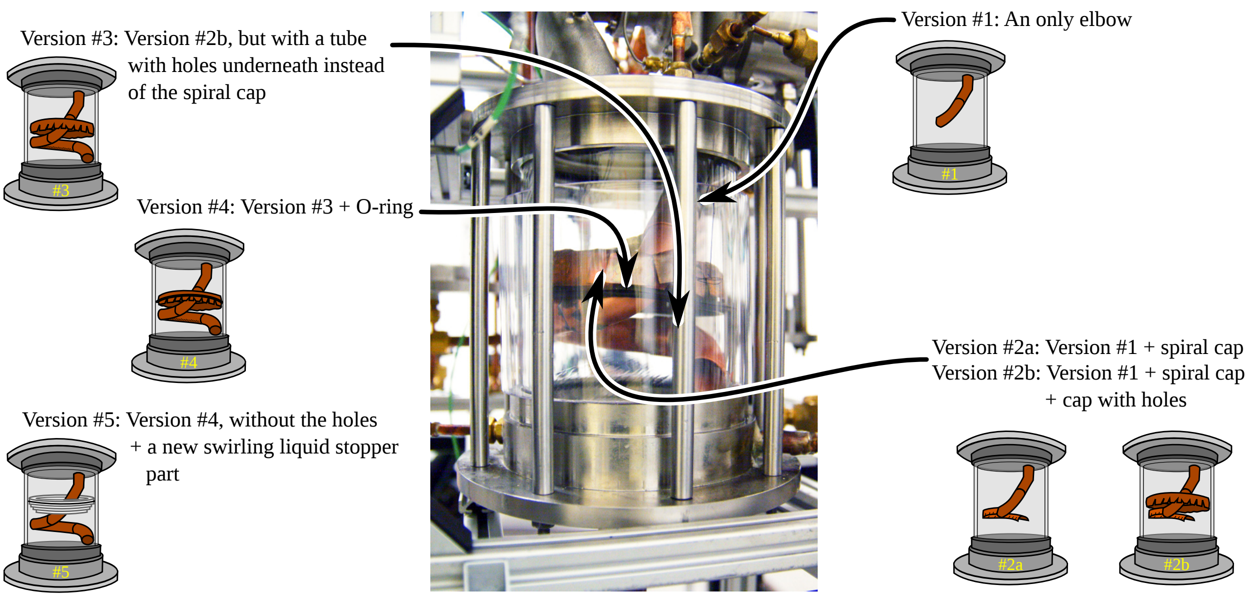


(a) Refrigeration diagram



(b) Entropy diagram

Figure 3: Thermodynamic diagrams for the OP A-7.0/W35.6. Component numbers are detailed in Fig. 2.



(a) Evolution of the economizer, from version #1 to version #5.

(b) Economizer version #5 in action.

Figure 4: Evolution of the economizer towards effectiveness

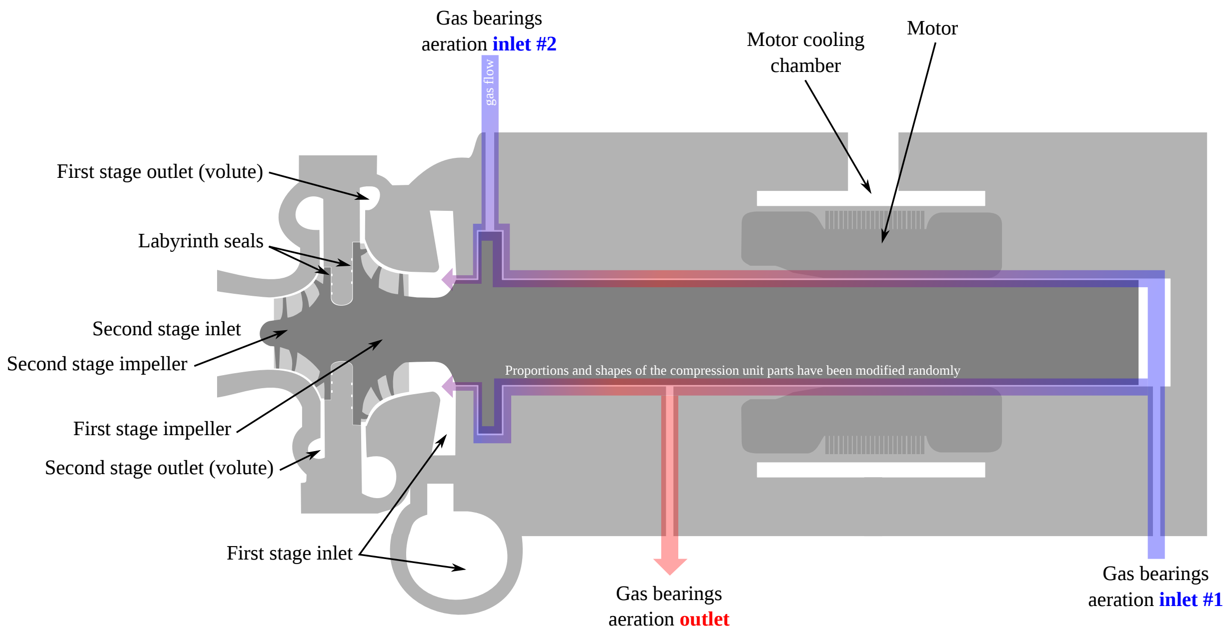


Figure 5: Compression unit schematics with correct auxiliary refrigerant circuits inlets and outlets.

Figure 6: The compression unit is equivalent to two 3kWe scroll compressors.

instead of 5 K (AFNOR, 2011b, tab. 12) (standard condenser inlet/outlet water for this OP are 30°C at the inlet and 35°C at the outlet), and the condenser inlet temperature was not exactly at 35.0°C, but at 35.6°C. Nonetheless, it appears that the AWP performs close to devices currently on the market, ranking at the same heating power, for a COP between 2.36 and 2.75 (source: Eurovent database (Eurovent Certification, 2010)). The AWP performance is a little lower than the devices it compares with (0.2 to 0.5 points lower). Reaching this level of performance is encouraging, since this first working prototype is far from being an optimized industrial version. Nonetheless it already ranks close to the models currently on the market. About 6% of the energy was dissipated in the environment during this experiment A-7.0/W35.6. Those 6% include the heat losses of the inverter (component #14). For the OP A-7/W35, COP was equal to 2.69 ± 0.03 (Eq. 1) while the exergy efficiency of the heat pump was equal to 32.5 ± 0.5 (Eq. 6), for a heat power at the condenser of 10656 ± 137 W. The exergy efficiency of the first compression stage was equal to 62 ± 37 (Eq. 2) while its isentropic efficiency was equal to 88 ± 11 (Eq. 4). The exergy efficiency of the second compression stage was equal to 59 ± 16 (Eq. 3) while its isentropic efficiency was equal to 82 ± 18 (Eq. 5). The motor exergy efficiency was equal to 91.92 % (Eq. 7). The inverter is considered to be inside the internal frontier in the model presented in Fig. 2, since its efficiency, not properly known, is assumed to be at a constant value of 85% (measured only once on the AWP). The COP is consequently defined as expressed in Eq. 1, and considers the inverter to be part of the internal system. COP does not take into account the pumps or the fan energy consumption for the auxiliary fluids.

$$\varepsilon_h = COP_h = \frac{\dot{Y}_h^-}{\dot{E}^+} = \frac{\dot{Y}_{3\to15}^-}{\dot{E}_{el\to14}^+} \tag{1}$$

$$\eta_{s,\,cp1} = \frac{\dot{M}_{29\to1}\,h_{1,\,out,\,s} - \dot{M}_{29\to1}\,h_{1,\,in} + \dot{Q}_{11\to1} - \dot{Q}_{1\to2}}{\dot{M}_{29\to1}\,h_{1,\,out} - \dot{M}_{29\to1}\,h_{1,\,in} + \dot{Q}_{11\to1} - \dot{Q}_{1\to2}} \tag{4}$$

$$\eta_{cp1,\,imp} = \frac{\dot{M}_{1\to25}\,\left(k_{1,\,out} - k_{1,\,in}\right)}{\dot{E}_{11\to1} - \dot{E}_{1\to2}} \tag{2}$$

$$\eta_{s,\,cp2} = \frac{\dot{M}_{28\to2}\,h_{2,\,out,\,s} - \dot{M}_{28\to2}h_{2,\,in} + \dot{Q}_{1\to2} - \dot{Q}_{2\to10}}{\dot{M}_{28\to2}\,h_{2,\,out} - \dot{M}_{28\to2}h_{2,\,in} + \dot{Q}_{1\to2} - \dot{Q}_{2\to10}} \tag{5}$$

$$\eta_{cp2,\,imp} = \frac{\dot{M}_{28\to2}\,\left(k_{2,\,out} - k_{2,\,in}\right)}{\dot{E}_{1\to2}} \tag{3}$$

$$\eta_{heatpump} = \frac{\dot{E}_{yh}^-}{\dot{E}_{el}^+} = \frac{\dot{M}_{17\to15}\,\left(k_{15,\,out} - k_{15,\,in}\right)}{\dot{E}_{el\to14}} \tag{6}$$

$$\eta_{motor} = \frac{\dot{E}_{13\to12} + \dot{M}_{3\to7}\,\left(k_{7,\,out} - k_{7,\,in}\right)}{\dot{E}_{14\to13}} \tag{7}$$

6. MAIN ISSUES ENCOUNTERED

6.1 Excess of compressor thrust forces during deceleration

When the compressor unit decelerates, the rotor speed decreases quite fast, since the rotor is mechanically loaded by the compressor work. In less than 2 seconds, the rotor speed drops below 100 krpm. Below 30 krpm, bearing touch-down occurs. Between 60 and 80 krpm, there is a dangerous operation zone, in particular if there is a high pressure level difference between the sources. As differences of pressure levels in the compression unit induce thrust forces on the axial bearing, and as both the bearings stiffness and load capacity increase with rotor speed, there are situations where the external force applied on the bearings might be too high for their nominal load capacity.

6.2 Significant increase of the temperatures in the labyrinth seal with rotation speed

A significant increase of the gas temperature in the labyrinth seal is observed for rotor speeds above 160 krpm. This observation is possible only through the use of the model presented in Fig. 2. Indeed, none of the temperatures of the compressor parts have been recorded; only the gas temperatures are known. Since only inlet and exhaust flow temperatures were measured, the gas temperatures inside the compressor unit are deduced from the model. The solving of this model allows to conclude that the order of magnitude of the shaft temperature in the thrust bearings area is of about 150°C, since the gas temperature approaches this temperature. The analysis allows to conclude that the hottest location in the whole compression unit and in the whole circuit is located in the axial bearing area, but the exact location in the bearings set is not known accurately.

6.3 Poor exergy efficiency of the evaporator and fluids maldistribution

The air ducting of the heat pump housing used as a base for the AWP design occurred to be designed with compactness in mind and has not been designed for optimum air stream or noise reduction. As a result, the air stream distribution in the air ducting is far from being satisfactory. Maldistribution of the refrigerant and/or of the air in the evaporator channels has been observed during the experiments. Differences of superheat at the outlet of the evaporator circuits have been observed as high as 8K. Consequently, in order to avoid liquid phase at the outlet of the evaporator, high overall superheat values at the inlet of the first stage compressor had to be imposed. The maldistribution observed in the refrigerant flow may be caused by 2 factors: (a) a non-ideal distribution of the liquid/gas flow coming from the lower stage expansion valve, or (b) a non-ideal distribution of the air flow in the air ducting implying uneven speed profiles in the evaporator sections.

6.4 Lubricant and particles pollution

Lubricant oil and particles pollution are dangerous for the compression unit which may break if exposed to that pollution. It is therefore advisable to avoid their presence in the circuits. No tolerance tests to that pollution has been performed so far. Oil pollution coming from the refrigeration bottles has been detected after the tests, notably in the evaporator and in the compression unit motor-cooling chamber, that act as oil-refrigerant separators.

6.5 Setting an appropriate flow rate in the auxiliary refrigerant circuits

The gas bearings aeration circuit includes a 0.5μm-filter which is by-far the main pressure drop of this auxiliary circuit, which means that the needle valve located before the filter has a low authority on the circuit flow rate. The filter is necessary to protect the gas-lubricated bearings from dust particles that could block grooves in the set of bearings. In order to decrease the filter pressure drop, increasing the surface of the filter or mounting more filters in parallel would have been valid solutions.

During the experiments, the flow in the motor cooling chamber was often too high. Ideally, the flow of refrigerant sent to the motor cooling circuit should be fully evaporated, which was not the case, most of the time. The oil separation problem is added to this flow regulation issue. Indeed, in the AWP, the motor cooling chamber is set in a pool-boiling configuration instead of flow-boiling configuration. Consequently, it acts as an oil-refrigerant separation device (lubricating oil pollution). The presence of this oil significantly decreases the performance of the motor cooling system, as the gap between the two walls of the motor cooling chamber is small, and because there is no circulation of the refrigerant around the motor. A defective motor cooling system can result in unexpected deformations of the motor parts, and especially of the shaft and can also result in an overheat of those parts, damaging them heavily. The mounting of the magnets on the shaft is also sensitive to overheat. Consequently, a flow boiling motor cooling system might be a better option for the motor cooling circuit until clean and lubricant-free refrigerant can be obtained and guaranteed by the industry supply chain.

6.6 Reaching a given OP

During the experiments, 2 additional phenomena could be observed:

- Absolute pressure in the economizer (component #8) was barely affected during the experiments by the settings of the prototype, including compressor speed and valves settings.
- The subcooling value was stabilizing itself around an almost fixed value and was almost independent of the setting of the second stage expansion valve.

The two phenomena are explained by the nature of the control system needed for such a heat pump: the intermediate pressure level can not be controlled independently and is a consequence of the choice of the compression unit rotation speed, the setting of the valves, and the compressor maps. Moreover, the temperature of the liquid phase in the economizer defines the pressure level of the economizer because of the thermal inertia of the liquid. Those observations lead to the conclusion that the control system needs to control the valves and the rotation speed all together and that the intermediate pressure level can not really be set at a chosen value directly, which may imply that inlet guides may be needed on one of the impellers inlets.

7. CONCLUSIONS

The feasibility and the potential of multistage oil-free variable-speed domestic heat pumps powered by twin-stage radial compressors has been demonstrated successfully. The OP A-7.0/W35.6 presented in this work demonstrates the feasibility of practical domestic heat pumps for space heating, as this OP is a typical domestic heat pump OP for floor heating technologies. Higher temperature lifts need to be reached and tested, but the feasibility of the concept has been demonstrated. The performance reached with the AWP is considered very promising and constitutes a breakthrough in the domain. The performance of the AWP, being a bit lower than the one of the devices on the market, with no optimization of the circuits design, with manual control, with a non-optimized refrigerant charge, and with numerous issues and avoidable energy losses, is already a significant achievement. Accounting for the numerous design issues which can be improved or solved in this first prototype, the performance is promising and shows that using radial compressors rotating on gas bearings technology for domestic heat pumps applications is already a success. Of course, the challenges are still numerous before domestic heat pumps equipped with radial compressors reach the market. The issues to be solved are substantial but paths of solutions have already been offered (Carré, 2015, p. 35–90). So far, no unsolvable problem could be identified. Defrosting and cycle inversion are still important possible issues, but strategies and design of key components like the economizer and the first stage separator already answer to a good share of the potential problems that could occur.

NOMENCLATURE

AWP	Air-Water heat pump Prototype.	
COP, ε	Coefficient Of Performance.	(-)
CTI	Swiss Commission for Technology and Innovation.	
$\dot{E}$	Work energy/exergy rate.	(W)
h	Specific enthalpy.	(J kg^{-1})
IEA19	IEA19 is a set of 19 countries where extended energy data is available. Notably those 19 countries have domestic sector final energy consumption statistics available.	
OP	Operating Point.	
k	Specific coenthalpy ($k = h - T_a s$).	(J kg^{-1})
$\dot{M}$	Mass flow rate.	(kg s^{-1})
η	Exergy efficiency.	(-)
s	Specific entropy.	(J kg^{-1} K^{-1})
T	Temperature.	(K)
$\dot{Y}$	Transformation energy rate.	(W)

Subscript

a	Atmosphere.
cp	Compressor, compression stage.
el	Electricity.
h	Heating, heat.
imp	Impeller, at the impeller level.
s	Isentropic.
y	Transformation (exergy).

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